



Experimental analysis of a micro-trigeneration system composed of a micro-cogenerator coupled with an electric chiller



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HIGHLIGHTS

- On-site performance of a micro-trigeneration plant was measured and analyzed.
- The plant was composed of a micro-cogeneration unit feeding an electric chiller.
- Data under both transient and steady-state operation were collected.
- The measured performances were compared with those of conventional systems.
- Primary energy, operating costs and carbon dioxide emissions were compared.

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ABSTRACT

Micro-trigeneration is one of the most promising technologies allowing the simultaneous production of electricity, heat and cooling on a small-scale basis, with a potential contribution to significant primary energy saving and greenhouse gas emission reduction if compared with the conventional separate energy production.

The on-site performance of a micro-trigeneration plant serving the Built Environment Control Laboratory of Second University of Naples (south of Italy) during summer were measured and analyzed; the system was composed of a 6.0 kW natural gas-fueled reciprocating internal combustion engine-based micro-cogeneration unit feeding a 7.5 kW electric air-cooled water chiller.

The experimental data were compared with the performance of conventional systems based on separate energy production in order to assess the feasibility of the proposed system; the comparison was performed according to the Italian scenario by estimating the potential savings in terms of primary energy consumption, operating costs and carbon dioxide equivalent emissions.

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1. Introduction

Population growth and technological advancement over the last two decades, along with the desire for higher life standards and comfort levels have led to an unprecedented increase in energy consumption worldwide [1]. Presently, this consumption is mainly due to the air-conditioning demand and is expected to grow considerably within the next few years [2]. A large amount (about 33%) of the overall world energy production is obtained by using oil, while the contribution of renewable energy resources is

still less than about 9% [3]; this heavy reliance on conventional fossil fuels has led to an increase in the global energy-related carbon dioxide emissions. The depletion of resources as well as environmental conscience regarding global warming and climate change have urged the need for a complete change in energy production, supply and consumption patterns in order to reduce both the energy demand and improve energy efficiency of the energy production systems; taking into account that buildings contribute about 30–40% of the overall energy consumption in all the industrialized countries, a big effort should be made mainly in this sector.

On the demand side, it is widely accepted that the improvement of the building envelope's insulation takes top priority. Wide-ranging options exist on the supply side for the provision of

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electricity and heat; among these options, micro-cogeneration is emerging as a fast growing technique to increase energy conversion efficiency in domestic and small-scale applications; this technology is defined as the local combined production of electrical and thermal energy from a single fuel source with an electric output lower than 50 kW_{el}, and its deployment is considered by the European Community as one of the most effective measures to save primary energy, reduce greenhouse gas emissions and contribute to the security of energy supply [4] with respect to the conventional separate production of electricity (in centralized power plants) and heat (in boilers). There are several technologies being developed for micro-cogeneration including internal combustion engines, gas and steam turbines, fuel cells and external combustion Stirling engines. Among these options, reciprocating internal combustion engine-based cogeneration systems are usually the best choice for residential applications thanks to their well-proven technology, robust nature and reliability [5–7].

In many countries there has been an increasing demand of cooling energy during the warm season generally satisfied by electrically-driven units [8]. This trend has involved an increase of the power generation capacity of electric utilities and contributed to electrical load peaking and subsequent network congestion and failure events in different power systems worldwide [9]. This has strengthened the awareness of governments, manufacturers and communities of energy issues, pushing forward the search for innovative systems for local energy generation. The presence of a threefold energy demand (namely, electricity, heat and cooling) leads to the possibility of profitably combining micro-cogeneration units with various technologies currently available for cooling generation (heat-fired or electricity-fired). Combination of MCHP (micro combined heat and power) systems to various thermally fed or electrically-driven cooling systems allows to set up a so-called micro combined cooling heat and power (MCCHP) system [6,10], that represents the production in-situ of a threefold energy vector requested by the user from a unique source of fuel (e.g., natural gas).

The processes and technologies for micro-trigeneration that are nowadays available can lead to a number of advantages [6,10,11]:

- promoting energy efficiency and use of renewable sources in alternative to the traditional separate generation;
- reducing the global energy-related carbon dioxide emissions;
- deferring huge investments in new large generation plants, substations or infrastructures;
- preferring the type of energy with higher local availability of the corresponding “fuel”;
- reducing both the energy dependency and vulnerability of the electrical system from the effects of grid congestion or service interruptions;
- developing competitive electricity markets and increasing the customer participation to the market, providing solutions to exploit the price elasticity to the electrical demand;
- the local resources connected to the electrical grid may provide benefits on the electrical network reliability and power quality.

Owners must consider a range of potential problems in tandem with the potential benefits of trigeneration, such as high capital costs, lack of policies, regulations and policy framework, shortages of investment finances [6].

Several thermally fed or electrically-driven cooling systems combinable with a micro-cogeneration unit can be considered: absorption chillers and heat pumps [12], engine-driven chillers and heat pumps [13], electric chillers and electric heat pumps. However, additional technologies combinable to MCHP systems can be adopted: in particular, adsorption [14] and solid or liquid desiccant

systems [15] are being more and more used; a combination with low temperature thermal sources such as solar power is also an attractive option.

Among the above-mentioned alternatives for composing tri-generation systems, the combination of MCHP units with electric chillers/heat pumps is one of the most promising taking into account that it allows to:

- run the cogeneration device for longer time at higher average loads, thus achieving better energy and environmental performance and shorter investment pay-back periods;
- reduce the electrical load peaking corresponding to the large cooling loads during summer and subsequent network congestion and failure events;
- profitably exploit the excess electrical production from renewable sources in case of selling the surplus electricity to the grid is not inconvenient.

Various studies relating to different countries [16–23] have highlighted that the incorporation of electric chillers/heat pumps to the traditional micro-cogeneration scheme represents an efficient and economical alternative to traditional systems. In particular, Cooper et al. [16] analyzed the relative energy and environmental performances of six micro-trigeneration systems composed of an electric air source heat pump and a solid oxide fuel cell micro-cogeneration unit by using a simulation approach. They found that the way in which the electric chillers/heat pumps are controlled has the potential to reduce their impacts by more than a third, while the largest savings achieved by the micro-cogeneration unit occur when it is running continuously at full output. Malinowska and Malinowski [17] compared a small-scale combined heat and power plant incorporating an electric heat pump with a conventional system by carrying out a theoretical parametric analysis in terms of exergetic efficiency; their work highlighted that the exergetic efficiency of the cogeneration plant can be over twice higher than that one of the corresponding conventional system. Mancarella [18] presented a novel theoretical approach to the energy and carbon dioxide emission modeling of cogeneration systems coupled to electric heat pumps, providing an indicative assessment of the benefits potentially brought with respect to conventional energy systems and highlighting the most relevant parameters involved in the analysis of such systems (namely, the efficiencies of both cogeneration unit and separate energy production, the coefficient of performance of electric heat pumps, the operational energy-shifting values, and the emission factors of the energy inputs). The experimental performances of a prototype plant consisting of a micro-cogeneration unit coupled with an electric heat pump installed in Vigo (North-West of Spain) were analyzed from an energy point of view by Porteiro et al. [19,20]. The experimental results highlighted that the best performance can be achieved during winter. Smith and Few [21,22] analyzed the steady-state operation of a domestic scale prototype micro-cogeneration plant incorporating an electric heat pump in terms of operational costs, first-law and second-law of thermodynamics; the experimental data clearly demonstrated the advantages of the investigated system with respect to a conventional plant from both energy and economic point of views. The results of prototype testing were further used by the authors in the development of a simulation model [23] and a good agreement with experimental results was found.

As highlighted by the literature review, there are few works evaluating the performance of a micro-cogeneration device feeding an electric chiller/heat pump; in addition several studies [16–18,23] were performed by using a theoretical/modeling approach (instead of an experimental analysis) or focusing on the

experimental steady-state operation [21,22]: these approaches neglect the dynamic response of the units due to the fact that, as a general rule, the losses associated to the start-up and shut-down cycles imply that the efficiency of energy conversion systems under real operating conditions is considerably lower than that associated to a full load operation. It can be also noticed that the environmental performance were evaluated only by Cooper et al. [16] and Mancarella [18], while the economic analysis was performed only by Smith and Few [21,22]. Based on the above considerations, it can be easily stated that the experimental performance of micro-trigeneration systems incorporating electric chillers/heat pumps need to be investigated under a wider range of operating conditions by using a more detailed approach.

In this paper, the in-situ performances of a micro-trigeneration plant composed of a 6.0 kW_{el} natural gas-fueled reciprocating internal combustion engine-based micro-cogeneration unit coupled with a 7.5 kW_{cool} electric air-cooled water chiller (ACWC) were measured and analyzed. The system provides electricity, heat and cooling to the Built Environment Control Laboratory of Second University of Naples, located in the south of Italy; the ACWC was linked to the MCHP side by using the so-called electrical bottoming (or series) generation mode, i.e. the electric chiller was cascaded to the topping cogeneration plant feeding the ACWC with the cogenerated electricity. The experimental data were collected during various days of last summer and compared with the performance of a conventional system based on separate energy production in order to assess its feasibility; the comparison was performed according to the Italian scenario from energy, economic and environmental point of views with the main aim of estimating the potential savings of the proposed plant in terms of primary energy consumptions, operating costs and global carbon dioxide emissions.

In Section 2, the experimental plant and related instrumentation are described in detail; a detailed uncertainty analysis is carried out in Section 3, while Section 4 reports the experimental results. Finally, in Section 5 both the methodology and results of the comparison between the proposed system and the conventional system assumed as a reference are highlighted and discussed in depth.

2. Experimental set-up

A schematic view of the experimental apparatus, detailing instrumentation, is shown in Fig. 1. The set-up is installed in the Built Environment Control Laboratory of Second University of Naples consisting of two floors with a floor area of about 600 m² and a total volume equal to 4200 m³; the building is located in Frignano, a municipality in the south of Italy (around 20 km from Naples), and it hosts undergraduate and graduate students, Ph.D. students, researchers and professors. The plant is composed of the following main components:

- > micro-cogeneration unit (MCHP);
- > air-cooled water chiller (ACWC);
- > hot water storage (HWS);
- > three fan-coils (FC).

Real photographs of the components of the test facility under investigation are reported in Fig. 2.

The micro-cogeneration device is the model GECC60A2 (NR-P) commercialized by the AISIN SEIKI company [24]; the main characteristics of the MCHP unit are reported in Table 1 based on manufacturer data. The system is based on a natural gas-fueled reciprocating internal combustion engine with three cylinders, four strokes and a total displacement of 952 cm³. The engine drives a synchronous generator to deliver electrical power up to a nominal value of 6 kW_{el}; at the same time, the cogeneration unit allows recovering the heat otherwise dissipated by the engine with a nominal thermal output of 11.7 kW_{th}. The system can operate under both electric and thermal load following logics:

- in the case of electric load following operation, the cogeneration system produces electricity by varying the power supplied through the inverter on the basis of the instantaneous user's power demand. Heat is recovered during unit operation, even if its level depends on the electric output of the device; the electric grid is used as back-up system;
- in the case of thermal load following operation, the unit operates according to an external temperature signal coming from a

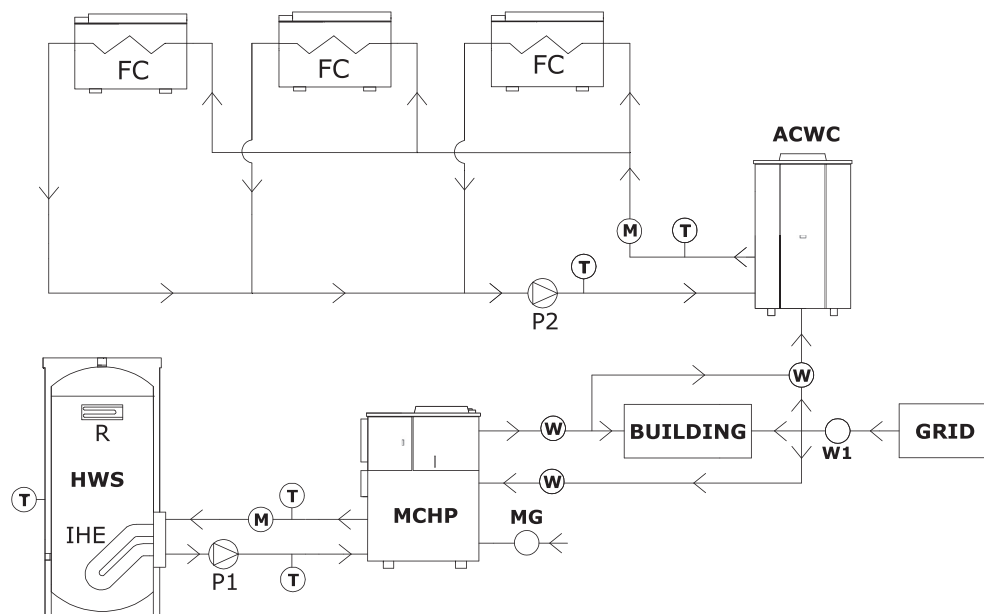


Fig. 1. Experimental apparatus.



Fig. 2. Photographs of the experimental facility under study: (a) HWS, (b) ACWC, (c) MCHP.

specific thermostat placed on the hot water storage: when that temperature is lower than the set-point value, the unit operates providing its maximum electric and thermal outputs; in the case of the water temperature in the tank exceeding the set-point value, the unit stops. Any unused excess of electric energy that is not self-used can be sent to the power line.

The experiments described in the next sections were performed by operating the MCHP unit under electric load following logic and its electric output was used to cover the electric requirements associated to the operation of the following main loads: ACWC, MCHP unit, pumps, lighting, PCs, printers, plotters, auxiliaries, etc. There is not any priority among the different electric loads.

In comparison to the thermal load following logic, the electric load following operation should increase the hours of operation of the cogeneration unit during summer thanks to the larger amount of electric energy required by the electric chiller for cooling purposes (mainly in the case of residential applications where the thermal requests for domestic hot water production are not very

relevant). This should enhance the potential savings associated to the utilization of the micro-trigeneration system composed of the MCHP unit and the ACWC with respect to the traditional systems based on separate energy production.

Thermal recovery is obtained from the exhaust gases and the engine jacket of the micro-cogenerator by using a water-glycol mixture as coolant; the recovered heat by the water-glycol mixture is transferred to the cooling water flowing through a high efficiency plate heat exchanger located inside the unit; the cooling water is circulated by the pump P1. The cogeneration system has an internal control to dissipate thermal energy by means of an internal air-cooled radiator (the radiator fan air flow is equal to 35 m³/min) and keep the internal water-glycol temperature lower than a fixed value (around 70 °C) in order to prevent conditions endangering the engine block. Scheduled maintenance of the MCHP unit is every 10,000 h, while the engine oil replacement is every 30,000 h; usual filters and sparking plugs replacement takes place every 10,000 h.

Additional details about the MCHP operation can be found in Refs. [25,26].

The recovered heat is stored by means of a tank characterized by a nominal internal volume equal to 1.0 m³ [27]; the hot water storage is insulated with a 100 mm flexible polyurethane layer (characterized by a thermal conductivity equal to 0.038 W/m K) and is furthermore equipped with an auxiliary 4.0 kW electric resistance. An internal heat exchanger, characterized by a heat transfer area equal to 2.0 m², is located in the lower part of the tank.

During winter the thermal energy stored in the tank is used for heating purposes and/or for domestic hot water production, while during summer it can be used to activate a three-phase absorption chiller and/or produce domestic hot water. In this paper the heat provided by the MCHP unit used only for domestic hot water production is investigated. The operation of the three-phase absorption chiller has not been investigated and hence not included in Figs. 1 and 2.

The electric power supplied by the cogeneration device is partly used for feeding the electric chiller. The investigated air-cooled water chiller is the model CRA/K/SP 25 commercialized by the company CLINT [28]. The unit has a nominal cooling capacity of 7.5 kW, with a nominal power of the compressor equal to 2.4 kW; it is equipped with a single rotary compressor and uses R410A as a refrigerant. The condenser is made of copper tubes and aluminum finned coil, complete with a moisture drain pan; the evaporator is a stainless steel braze-welded plates type, insulated with flexible closed cells material; the fan is a double inlet centrifugal type directly driven by a single-phase electric motor, with a nominal air volumetric flow rate of 0.94 m³/s. The refrigerant circuit, in copper tubing, includes a manual reset high pressure switch, dryer filter, expansion valve; the water circuit, in copper tubing, includes a storage tank (0.05 m³), water gauge, safety water valve, expansion vessel (0.002 m³), electrical pump P2, water differential pressure switch, manual air release valve, plant charge and discharge shut off valve. The refrigerant flowing into the ACWC thermally interacts with water at the evaporator side, and it condensed by means of an air flow at condenser side. The main characteristics of the air-cooled water chiller are summarized in Table 2 based on manufacturer data.

The water cooled by the ACWC is moved into three fan-coils installed in order to control the indoor air temperature during the summer within a portion of the whole laboratory. The model FVW-FIV-32 commercialized by the company CLINT [28] is used as fan-coils: each unit is made of copper pipes and aluminum fins, with a nominal cooling output of 2.02 kW (in the case of ambient air temperature equal to 27 °C and inlet/outlet water temperature of 7/12 °C), volumetric air flow rate ranging from 250 up to 430 m³/h and nominal electric consumption in the range 0.02–0.05 kW.

Table 1
Main characteristics of the MCHP unit based on manufacturer data [24].

Model	GECC60A2 (NR-P)
Engine type	Reciprocating internal combustion engine, water cooled, 4 cycles, 3 cylinders
Displacement (cm ³)	952
Speed revolution (rpm)	1600–1800
Fuel	Natural gas or LPG
Generator type	Permanent – magnet type, synchronous generator 16 poles
Rated electric output (kW)	0.3–6
Heat recovery rate (kW)	11.7
Operating sound at 1.0 m distance and 1.5 m height (dB)	54
Electric efficiency at maximum load (%)	28.8
Thermal efficiency at maximum load (%)	56.2

Table 2

Main characteristics of the ACWC based on manufacturer data [28].

Model	CRA/K/SP 25
Refrigerant	R410A
Refrigerant charge (kg)	2.3
Compressor quantity/type	1/rotary
Condenser type	Copper tubes and aluminum finned coil
Evaporator type	Stainless steel braze-welded plates type
Fan type	Double inlet centrifugal type
Storage tank volume (dm ³)	50
Expansion vessel volume (dm ³)	2
Nominal cooling capacity ^a (kW)	7.5
Compressor nominal power ^a (kW)	2.4
Fan nominal power ^a (kW)	0.25
Pump nominal power ^a (kW)	0.2
Sound pressure level in free field conditions at 1 m from the unit and at 1.5 m from the ground (dB)	58

^a Ambient air temperature = 35 °C; evaporator water temperature inlet/outlet = 12/7 °C.

Additional details about the experimental plant layout and operation can be found in Refs. [29,30].

The experimental apparatus is well instrumented in order to measure the following parameters:

- temperature T_{air} of air within the room (communicating with the outside) where the ACWC is installed;
- temperature $T_{\text{w,HWS}}$ of water within the tank;
- volumetric flow rate of natural gas $\dot{V}_{\text{ng,MCHP}}$ entering the MCHP device;
- temperature of cooling water entering ($T_{\text{cw,MCHP,in}}$) and exiting ($T_{\text{cw,MCHP,out}}$) the MCHP device;
- volumetric flow rate of cooling water $\dot{V}_{\text{cw,MCHP}}$ entering the MCHP unit;
- temperature of cooling water entering ($T_{\text{cw,ACWC,in}}$) and exiting ($T_{\text{cw,ACWC,out}}$) the ACWC;
- volumetric flow rate of cooling water $\dot{V}_{\text{cw,ACWC}}$ entering the ACWC;
- electric power $P_{\text{el,MCHP,out}}$ supplied to the end-user by the MCHP device;
- electric power $P_{\text{el,MCHP,in}}$ consumed by the MCHP device during stand-by, warm-up and cool-down operation;
- electric power $P_{\text{el,grid,out}}$ supplied by the central grid;
- electric power $P_{\text{el,ACWC}}$ consumed by the ACWC.

Table 3 summarizes the main characteristics of the plant instrumentation.

Based on the above-mentioned directly measured parameters, the thermal power $P_{\text{th,MCHP}}$ recovered from the MCHP device, the primary power $P_{\text{p,MCHP}}$ consumed by the MCHP device, the electric efficiency $\eta_{\text{el,MCHP}}$ of the MCHP device, the thermal efficiency $\eta_{\text{th,MCHP}}$ of the MCHP device, the cooling output $P_{\text{cool,ACWC}}$ of the ACWC and the coefficient of performance COP_{ACWC} of the ACWC unit are calculated as follows:

$$P_{\text{th,MCHP}} = \dot{V}_{\text{cw,MCHP}} \cdot \rho_{\text{cw}} \cdot c_{\text{cw}} \cdot (T_{\text{cw,MCHP,out}} - T_{\text{cw,MCHP,in}}) \quad (1)$$

$$P_{\text{p,MCHP}} = \dot{V}_{\text{ng,MCHP}} \cdot \rho_{\text{ng}} \cdot \text{LHV}_{\text{ng}} \quad (2)$$

$$\eta_{\text{el,MCHP}} = P_{\text{el,MCHP}} / P_{\text{p,MCHP}} \quad (3)$$

Table 3

Main characteristics of the plant instrumentation.

Parameter	Instrument	Symbol in Fig. 1	Operating range	Uncertainty
$T_{\text{air}}, T_{\text{w,HWS}}, T_{\text{cw,MCHP,in}}, T_{\text{cw,MCHP,out}}, T_{\text{cw,ACWC,in}}, T_{\text{cw,ACWC,out}}, \dot{V}_{\text{cw,MCHP}}, \dot{V}_{\text{cw,ACWC}}$	Resistance thermometer Pt100	T	−50–100 °C	±0.2 °C
$\dot{V}_{\text{ng,MCHP}}$	Ultrasonic volumetric flow meter	M	0–50 l/min	±2.5% of full scale
$\dot{V}_{\text{ng,MCHP}}$	Thermal volumetric flow meter	MG	0–5.0 Nm ³ /h	±0.8% of reading ±0.2% of full scale
$P_{\text{el,MCHP,out}}, P_{\text{el,MCHP,in}}, P_{\text{el,ACWC}}$	Wattmeter	W	0–6 kW	0.2% of full scale
$P_{\text{el,grid,out}}$	Wattmeter	$W1$	0–10 kW	0.2% of full scale

$$\eta_{\text{th,MCHP}} = P_{\text{th,MCHP}} / P_{\text{p,MCHP}} \quad (4)$$

$$P_{\text{cool,ACWC}} = \dot{V}_{\text{cw,ACWC}} \cdot \rho_{\text{cw}} \cdot c_{\text{cw}} \cdot (T_{\text{cw,ACWC,in}} - T_{\text{cw,ACWC,out}}) \quad (5)$$

$$\text{COP}_{\text{ACWC}} = P_{\text{cool,ACWC}} / P_{\text{el,ACWC}} \quad (6)$$

The specific heat and density of cooling water, respectively, were assumed equal to $c_{\text{cw}} = 4.187 \text{ kJ}/(\text{kg K})$ and $\rho_{\text{cw}} = 1000 \text{ kg}/\text{m}^3$; the density and Lower Heating Value of natural gas, respectively, were considered equal to $\rho_{\text{ng}} = 0.72 \text{ kg}/\text{m}^3$ and $\text{LHV}_{\text{ng}} = 49,599 \text{ kJ}/\text{kg}$ according to the natural gas composition suggested in Ref. [31].

The signals coming from the resistance thermometers Pt100 are acquired by three cFP-RTD-124 analog input modules (commercialized by National Instruments® [32]), while the signals coming from the other sensors are managed by two cFP-AI-110 analog input modules (commercialized by National Instruments® [33]). Each acquisition device is a 16-bit resolution system with eight current outputs (4–20 mA). The digital data coming from the modules are sent to a personal computer; the software LabView 8.2 is used to set the acquisition frequency as well as monitor and/or record the measured and calculated parameters.

3. Uncertainty analysis

In this paper, the uncertainty associated to the parameters defined by Eqs. (1)–(6) are evaluated according to the procedure suggested by the Guide to the expression of Uncertainty in Measurement [34]. According to Ref. [34], the combined standard uncertainty u associated to a measurement Y determined from a number of input quantities $X_1, \dots, X_N$ can be calculated by using the following formula:

$$u(y) = \sqrt{\sum_{i=1}^N (\partial f / \partial x_i)^2 \cdot u^2(x_i)} \quad (7)$$

where f is the functional relationship between Y and $X_1, \dots, X_N$ and y is an estimation of the measure and Y obtained by substituting the measured values $x_1, \dots, x_N$ of the parameters $X_1, \dots, X_N$ in Equation (7).

The combined standard relative uncertainty of $P_{th,MCHP}$ is calculated as follows:

$$u(P_{th,MCHP})/P_{th,MCHP} = \sqrt{u^2(\dot{V}_{cw,MCHP})/\dot{V}_{cw,MCHP}^2 + u^2(\rho_{cw})/\rho_{cw}^2 + u^2(c_{cw})/c_{cw}^2 + u^2(\Delta T_{cw,MCHP})/\Delta T_{cw,MCHP}^2} \quad (8)$$

where $\Delta T_{cw,MCHP} = T_{cw,MCHP,out} - T_{cw,MCHP,in}$.

The cooling water volumetric flow rate is measured by an ultrasonic volumetric flow meter characterized by an uncertainty equal to $\pm 2.5\%$ of full scale (Table 3). In this paper, the uncertainty associated to both c_{cw} and ρ_{cw} is neglected, while the uncertainty associated to the temperature difference ΔT_{cw} is calculated as follows:

$$u(\eta_{el,MCHP})/\eta_{el,MCHP} = \sqrt{u^2(P_{el,MCHP})/P_{el,MCHP}^2 + u^2(P_{p,MCHP})/P_{p,MCHP}^2} \quad (11)$$

$$u(\Delta T_{cw,MCHP}) = \sqrt{u^2(T_{cw,MCHP,out}) + u^2(T_{cw,MCHP,in})} \quad (9)$$

where the uncertainty associated to the temperature measurement is reported in Table 3.

Our laboratory is not equipped with an instrument for measuring the natural gas composition: as a consequence, in the following, a relative uncertainty value equal to $\pm 5.0\%$ is assumed for the fuel lower heating value, according to what is reported in Refs. [36–38].

The combined standard relative uncertainty of $\eta_{el,MCHP}$ is calculated as follows:

where the uncertainty associated to $P_{el,MCHP}$ is reported in Table 3, while the combined standard relative uncertainty of $P_{p,MCHP}$ is defined by Eq. (10).

The combined standard relative uncertainty of $\eta_{th,MCHP}$ is calculated as follows:

$$u(\eta_{th,MCHP})/\eta_{th,MCHP} = \sqrt{u^2(P_{th,MCHP})/P_{th,MCHP}^2 + u^2(P_{p,MCHP})/P_{p,MCHP}^2} \quad (12)$$

The combined standard relative uncertainty of $P_{p,MCHP}$ is obtained from the following equation:

$$u(P_{p,MCHP})/P_{p,MCHP} = \sqrt{u^2(\dot{V}_{ng,MCHP})/\dot{V}_{ng,MCHP}^2 + u^2(\rho_{ng})/\rho_{ng}^2 + u^2(LHV_{ng})/LHV_{ng}^2} \quad (10)$$

Fuel volumetric flow rate measurement is carried out by means of a thermal mass flow meter with an uncertainty rated by the manufacturer equal to $\pm 0.8\%$ of reading $\pm 0.2\%$ of the full scale (Table 3). The accuracy in density measurement depends mainly on the uncertainty related to the mixture composition [35] and the

where the combined standard relative uncertainty of $P_{th,MCHP}$ is obtained from Eq. (8) and the combined standard relative uncertainty of $P_{p,MCHP}$ is calculated by using Eq. (10).

The combined standard relative uncertainty of $P_{cool,ACWC}$ is obtained from the following formula:

$$u(P_{cool,ACWC})/P_{cool,ACWC} = \sqrt{u^2(\dot{V}_{cw,ACWC})/\dot{V}_{cw,ACWC}^2 + u^2(\rho_{cw})/\rho_{cw}^2 + u^2(c_{cw})/c_{cw}^2 + u^2(\Delta T_{cw,ACWC})/\Delta T_{cw,ACWC}^2} \quad (13)$$

estimated value is typically around $\pm(0.10\text{--}0.50)\%$ of the reading [36]; in the following, a relative uncertainty equal to $\pm 0.10\%$ of the reading is assumed for the natural gas density.

where $\Delta T_{cw,ACWC} = T_{cw,ACWC,in} - T_{cw,ACWC,out}$.

The cooling water volumetric flow rate is measured by an ultrasonic volumetric flow meter characterized by an uncertainty

equal to $\pm 2.5\%$ of the full scale (Table 3). In this paper, the uncertainty associated to both c_{cw} and ρ_{cw} is neglected, while the uncertainty associated to the temperature difference $\Delta T_{cw,ACWC}$ is calculated as follows:

$$u(\Delta T_{cw,ACWC}) = \sqrt{u^2(T_{cw,ACWC,in}) + u^2(T_{cw,ACWC,out})} \quad (14)$$

where the uncertainty associated to the temperature measurement is reported in Table 3.

The combined standard relative uncertainty of COP_{ACWC} is obtained as follows:

$$u(COP_{ACWC})/COP_{ACWC} = \sqrt{u^2(P_{cool,ACWC})/P_{cool,ACWC}^2 + u^2(P_{el,ACWC})/P_{el,ACWC}^2} \quad (15)$$

where the uncertainty associated to $P_{el,ACWC}$ is reported in Table 3, while the combined standard relative uncertainty of $P_{cool,ACWC}$ is defined by Eq. (13).

4. Experimental results

In this section, the experimental results are described and discussed in detail.

Three experiments were carried out by operating the MCHP device under electric load-following logic:

- > test 1 (date: 26/06/2013, starting time: 14:13, ending time: 15:56);
- > test 2 (date: 05/07/2013, starting time: 12:09, ending time: 13:30);
- > test 3 (date: 05/05/2013, starting time: 9:10, ending time: 10:48).

Taking into consideration that the performance of the MCHP unit strongly depend on the electric load sensed by the device, three different tests characterized by three different levels of the electric energy consumed by the electric appliances (pumps, lighting, PCs, printers, plotters, auxiliaries) were performed. In particular, the electric demand associated to the electric appliances was minimum (maximum around 0.2 kW) during test 1 and maximum (ranging from around 2.0 kW up to around 2.4 kW) during test 3; it is almost constant (around 0.8 kW) during test 2.

The experimental data were recorded every 10 s during the tests 1–2, and every 30 s during the test 3 (the acquisition time is much lower than the time required by the MCHP unit to reach steady-state operation).

Fig. 3a–c shows the primary power $P_{p,MCHP}$ consumed by the MCHP unit (Eq. (2)), the electric power $P_{el,MCHP,out}$ supplied by the MCHP unit, the thermal power $P_{th,MCHP}$ recovered from the MCHP unit (Eq. (1)), the cooling output $P_{cool,ACWC}$ of the ACWC (Eq. (5)), the electric power $P_{el,ACWC}$ consumed by the ACWC, the electric power $P_{el,grid,out}$ supplied by the grid and the electric power $P_{el,MCHP,in}$ consumed by the MCHP unit measured during tests 1, 2, and 3, respectively.

The power consumed by electric appliances (pumps, lighting, PCs, printers, plotters, auxiliaries) can be derived from the experimental values reported in Fig. 3a–c as difference between the electric power supplied by both MCHP unit and electric grid ($P_{el,MCHP,out} + P_{el,grid,out}$) and the electric power consumed by both ACWC and cogeneration device ($P_{el,ACWC} + P_{el,MCHP,in}$). The power

consumed by electric appliances is very low during the test 1 (maximum around 0.2 kW), while it ranges from around 2.0 kW up to around 2.4 kW during test 3; it is almost constant (around 0.8 kW) during test 2.

These figures highlight that the air-cooled water chiller is operated under on/off control logic: it provides cooling energy only when the indoor air temperature is greater than the set-point value; as soon as the room temperature is equal to the set-point temperature, the ACWC switches in standby mode operation and the cooling output becomes zero. The cooling output of the air-cooled water chiller varies up to about 6.1 kW, while its maximum electric consumption is almost constant and equal to

about 2.7 kW. The trends of $P_{cool,ACWC}$ show that the output of the ACWC increases with the operation time.

The micro-cogeneration device exhibits four other operating modes with markedly different characteristics:

- stand-by: the unit is assumed to be in stand-by mode when no electrical or thermal output is requested; the MCHP device consumes around 90 W of electricity during the stand-by operation due to the operation of electronic controllers and auxiliary electrical systems within the unit;
- warm-up: the warm-up is the period during in which there is a static time delay between the activation of the unit (i.e. when the electric demand is larger than around 380 W) and electrical power generation; the MCHP consumes around 190 W of electricity during the warm-up due to the operation of electronic controllers and auxiliary electrical systems within the unit;
- cool-down: the cool-down period begins when the fuel supply has stopped and ends when the stand-by mode is entered; during the cool-down the MCHP unit is able to recover thermal power by consuming around 190 W of electricity and some natural gas (due to the operation of electronic controllers and auxiliary electrical systems within the unit);
- normal mode: during normal operation mode the MCHP system produces heat and power by varying the power supplied through the inverter on the basis of the user's power demand.

With respect to the last point, it should be noted that the cogeneration device is not capable of operating as a stand-alone unit; the unit shuts down immediately when the external power is lost, for safety reasons. This is the reason why:

- some electricity (around 380 W) is imported from the grid also during normal mode operation with the electric output of the MCHP device lower than its maximum electric output;
- the MCHP unit starts only in the case of the electric demand is larger than around 380 W.

Fig. 3a–c shows that the thermal transient behavior of the MCHP unit is not negligible; in fact, $P_{th,MCHP}$ slowly increases before reaching its steady-state value; the cogeneration system attains steady-state operation only after several minutes of operation. In particular, the experimental results highlighted that thermal output of the MCHP unit reaches about 90% of the steady-state value after around 600–650 s of operation with thermal transient becoming less and less relevant at increasing the electric load

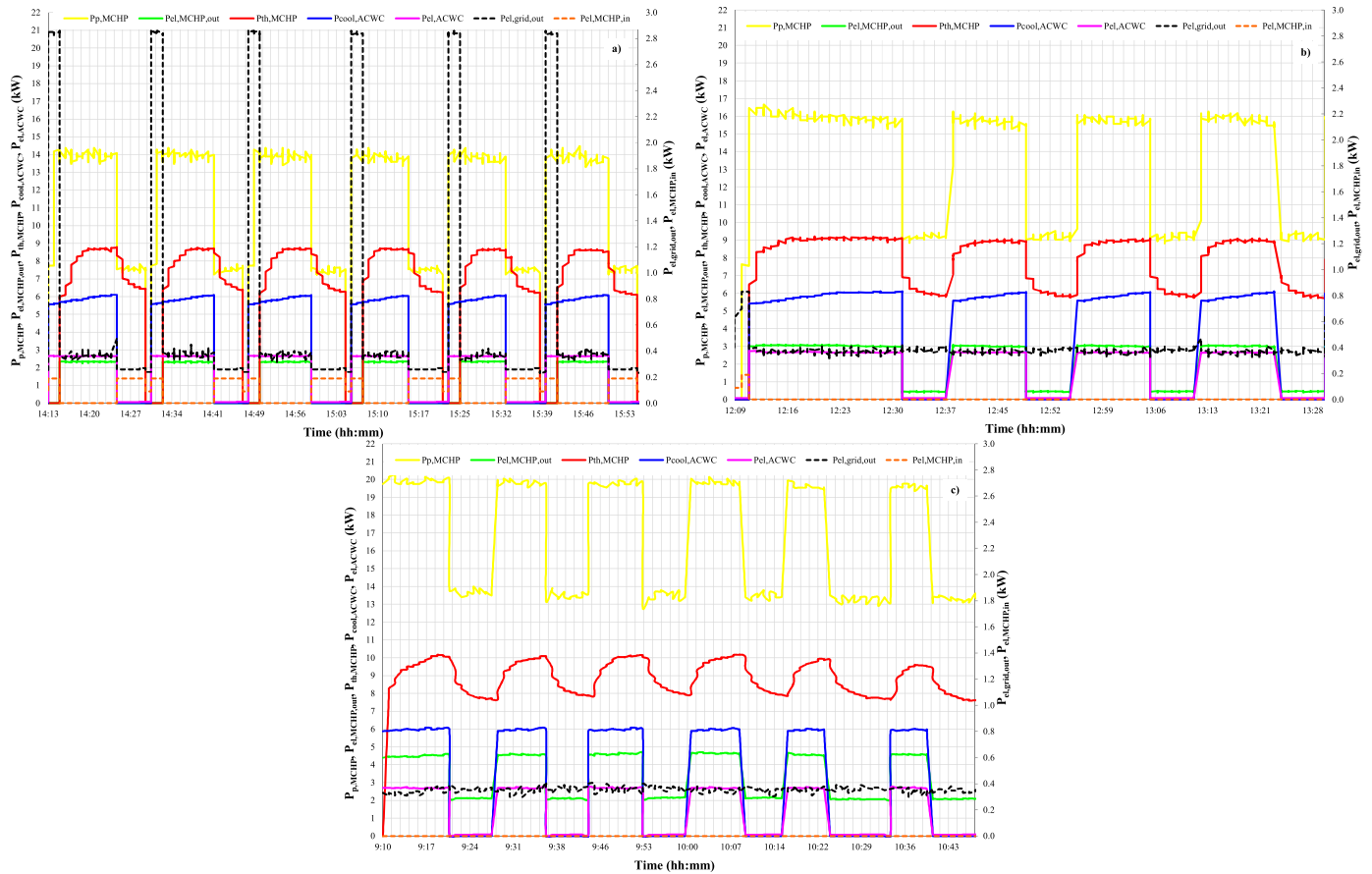


Fig. 3. Values of $P_{p,MCHP}$, $P_{el,MCHP,out}$, $P_{th,MCHP}$, $P_{cool,ACWC}$, $P_{el,ACWC}$, $P_{el,grid,out}$, $P_{el,MCHP,in}$ measured during test 1 (a), test 2 (b) and test 3 (c).

sensed by the cogeneration device; the measured data reported in Ref. [26] highlight the same characteristics about the thermal transient of the MCHP unit under investigation. The temperature of water entering the micro-cogeneration unit always remains lower than the maximum value allowed by the control logic of the MCHP device (around 70 °C) and, therefore, thermal energy cogenerated by the system was never dissipated during the experiments through the internal radiator. The thermal recovery behavior described above is related to the fact that the unit's management system allows heat recovery only when the water–glycol mixture temperature is higher than a given value: in case of its temperature level is lower than the fixed minimum value, the solution doesn't cross the plate heat exchanger, where the heat exchange between the cooling water and the water-glycol mixture should take place. The solution starts flowing, partially or totally, through the plate heat exchanger just in case of its temperature becomes enough high.

During the experiments the electric chiller operates under transient conditions due to the fact that the temperature of water entering/exiting the unit is not constant. In particular, during transient operation the temperature of water entering the ACWC varies from around 10.6 °C up to around 27.2 °C, while the cooling water exits the air-cooled chiller with a temperature in the range 5.8–27.2 °C.

Fig. 4a–c reports the electric efficiency $\eta_{el,MCHP}$ of the MCHP device (Eq. (3)), the thermal efficiency $\eta_{th,MCHP}$ of the MCHP device (Eq. (4)), and the coefficient of performance COP_{ACWC} of the ACWC (Eq. (6)) measured during tests 1, 2, and 3, respectively. The experimental data show that:

- both the electrical and thermal efficiencies of the micro-cogeneration unit are affected only by the MCHP power output: in particular, whenever the cooling fluid temperature and flow rate, $\eta_{el,MCHP}$ rises upon increasing the value of $P_{el,MCHP,out}$, while $\eta_{th,MCHP}$ increases upon decreasing the value of $P_{el,MCHP,out}$;
- for a given electric load sensed by the cogeneration device, the time variation of the electric efficiency is almost negligible;
- for a given electric output of the MCHP unit, the value of thermal efficiency during steady-state operation is completely different from that one observed at the beginning of the tests (this experimental behavior demonstrates that taking into account the thermal transient behavior of the unit is mandatory in assessing its performance);
- there is a negligible influence of the cooling water temperature on either the MCHP electrical or thermal efficiencies;
- the values of $\eta_{el,MCHP}$ vary up to around 24%, while the maximum value of $\eta_{th,MCHP}$ is around 78%;
- the values of COP_{ACWC} increase with the operation time;
- for a given value of external temperature T_{air} , the coefficient of performance of the ACWC increases with the values of $T_{cw,ACWC,out}$;
- the maximum value of COP_{ACWC} during transient operation is around 2.3.

Table 4 indicates the maximum values of the combined standard relative uncertainties associated to the parameters calculated by using Eqs. (1)–(6) based on data measured during tests 1, 2, and 3. This table highlights that $P_{cool,ACWC}$ and COP_{ACWC} are characterized

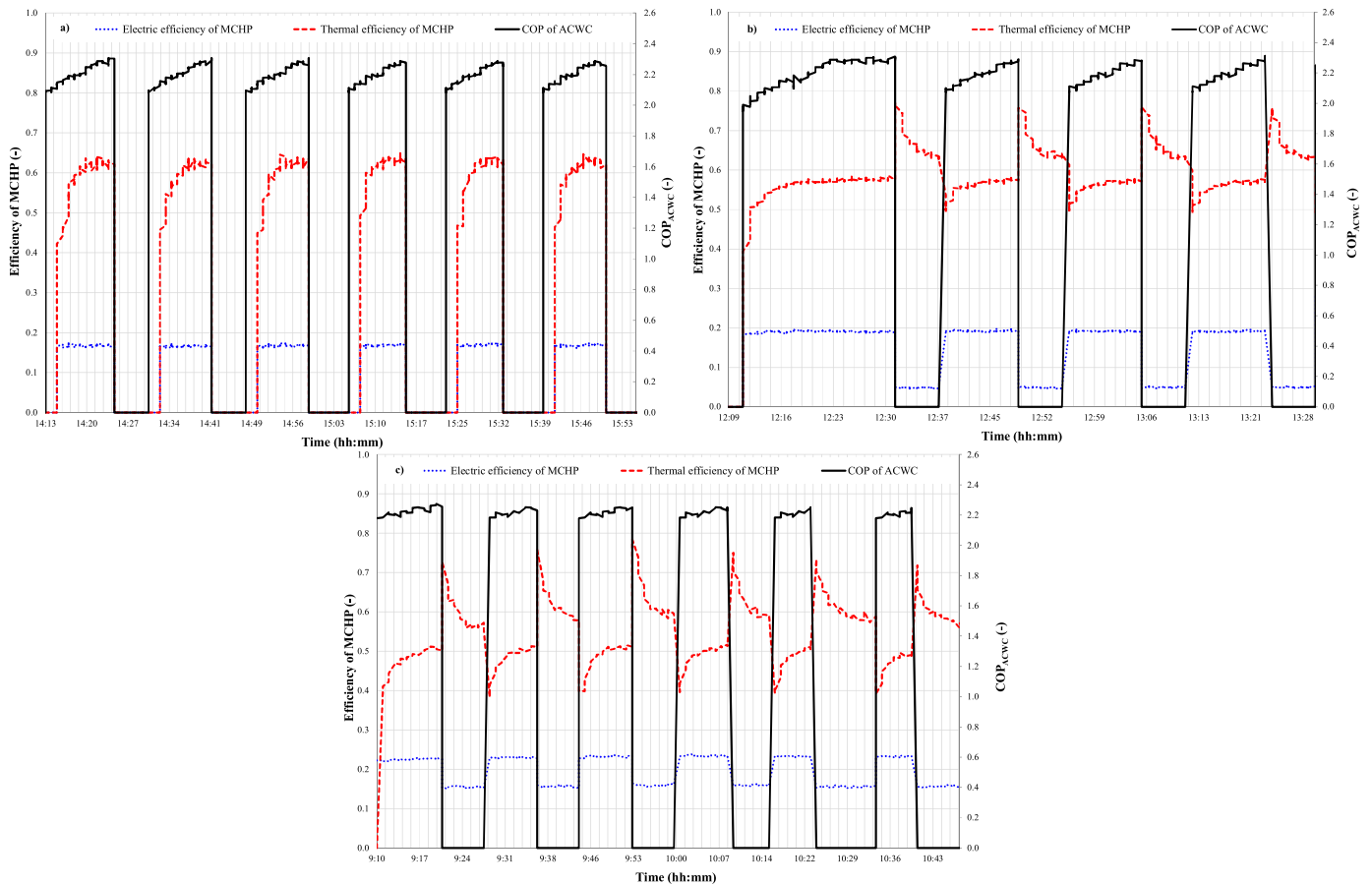


Fig. 4. Values of $\eta_{el,MCHP}$, $\eta_{th,MCHP}$ and COP_{ACWC} measured during test 1 (a), test 2 (b) and test 3 (c).

by the largest relative uncertainty, ranging from 13% up to 14%; the lowest relative uncertainties are those related to the $\eta_{el,MCHP}$ and $P_{p,MCHP}$, ranging from 2% up to 4%.

The maximum experimental value of $\eta_{th,MCHP}$ (around 78%) is larger than the thermal efficiency of the MCHP unit suggested by the manufacturer (56.2%). This is mainly due to the fact that $\eta_{th,MCHP}$ increases at decreasing the electric output of the cogeneration device: the maximum thermal efficiency of the MCHP system measured in this paper has been obtained by operating the system under a very low electric load (around 0.5 kW), while the value of $\eta_{th,MCHP}$ reported in Table 1 relates to the case with maximum electric load (6 kW). In addition the difference is related to the value of Lower Heating Value (LHV) used in thermal efficiency calculations: the LHV value used in this paper is 49,599 kJ/kg according to the natural gas composition suggested in Ref. [31], while the value used by the manufacturer is not clearly specified.

The difference between the maximum value of $\eta_{el,MCHP}$ measured in this paper (around 24%) and the value suggested by

the manufacturer (28.8%) is mainly related to the fact that $\eta_{el,MCHP}$ increases at increasing the electric output of the cogeneration device: the maximum experimental electric efficiency of the MCHP unit has been measured by operating the system under an electric load of around 4.5 kW, while the value of $\eta_{el,MCHP}$ reported in Table 1 relates to the case with maximum electric load (6 kW).

5. Energy, economic and environmental analysis: methodology

The measured performance of the system composed of the micro-cogeneration device and electric chiller are compared with those associated to a reference system based on separate energy production consisting of:

- a natural gas fired-boiler (for thermal energy production) with thermal efficiency η_B equal to 0.9;
- a power plant mix connected to the electric grid (for electricity production) with an efficiency η_{PP} equal to 0.461 (this value represents the power plant average efficiency in Italy, including transmission losses [39]);
- the same electric chiller experimentally investigated (for cooling energy production).

The comparison is performed in order to assess the feasibility of the proposed system and evaluate its potential benefits with respect to the system assumed as reference.

Several methodologies for the performance assessment of cogeneration and trigeneration systems have been proposed

Table 4
Maximum combined standard relative uncertainties associated to the parameters not directly measured.

	Test number 1	Test number 2	Test number 3
$u(P_{th,MCHP})/P_{th,MCHP}$	13%	12%	9%
$u(P_{p,MCHP})/P_{p,MCHP}$	2%	4%	2%
$u(\eta_{el,MCHP})/\eta_{el,MCHP}$	2%	4%	2%
$u(\eta_{th,MCHP})/\eta_{th,MCHP}$	13%	12%	9%
$u(P_{cool,ACWC})/P_{cool,ACWC}$	14%	14%	13%
$u(COP_{ACWC})/COP_{ACWC}$	14%	14%	13%

[40–42]. In this paper the analysis is carried out from energy, economic and environmental point of views by assuming that both systems have to satisfy the same electric, thermal and cooling requirements of the end-users. This section describes the methodology used in this work, while the results of the comparison are described and discussed in the next section.

The energy comparison is performed by using the Primary Energy Savings (PES) index, which is defined as follows:

$$PES = \frac{(P_p^{CS} - P_p^{PS})}{P_p^{CS}} \quad (16)$$

where P_p^{PS} is the total primary power consumed by the proposed system and P_p^{CS} is the total primary power consumed by the conventional system. According to its definition, a positive value of PES indicates that the system based on micro-cogeneration technology allows for a primary energy saving in comparison to the reference system.

The values of P_p^{PS} are obtained by means of the following formula:

$$P_p^{PS} = P_{p,MCHP} + P_{el,grid,out} / \eta_{PP} \quad (17)$$

where $P_{p,MCHP}$ is calculated by Eq. (2) based on the experimental results, and $P_{el,grid,out}$ is directly measured.

The values of P_p^{CS} are calculated as reported below:

$$P_p^{CS} = P_{th,MCHP} / \eta_B + P_{el,MCHP,out} / \eta_{PP} + P_{el,grid,out} / \eta_{PP} \quad (18)$$

where $P_{th,MCHP}$ is calculated by Eq. (1) based on the collected data, and both $P_{el,MCHP,out}$ and $P_{el,grid,out}$ are directly measured.

The energy comparison allows to estimate the potential savings in terms of primary energy consumption that can be achieved by adopting the MCHP device. However micro-cogeneration technology requires larger investments for the user when compared to the conventional separate energy production. In particular, the capital cost of the MCHP unit under investigation in this paper is around 18,000 €, the investment for the ACWC unit is around 2700 €, and the price of the hot water storage is around 1200 €. Taking into account the relevant capital cost associated to the cogeneration unit, positive net benefits are the mandatory condition for MCHP systems to be financially feasible. This is the reason why the comparison between the proposed and conventional systems is also performed in terms of operating costs by calculating the following parameter ΔOC :

$$\Delta OC = (OC^{CS} - OC^{PS}) / OC^{CS} \quad (19)$$

where OC^{PS} is the operating cost associated to the proposed system and OC^{CS} is the operating cost associated to the conventional system assumed as a reference. According to its definition, a positive value of ΔOC indicates that the system based on micro-cogeneration technology allows to reduce the operating costs in comparison to the conventional system.

The values of OC^{PS} are obtained by means of the following formula:

$$OC^{PS} = \dot{V}_{ng,MCHP} \cdot UC_{ng,MCHP} + P_{el,grid,out} \cdot UC_{el} \quad (20)$$

where $UC_{ng,MCHP}$ is the unit cost of natural gas consumed by the MCHP device, and UC_{el} is the unit cost of the electric energy purchased from the grid.

The values of OC^{CS} are calculated as reported below:

$$OC^{CS} = \dot{V}_{ng,B} \cdot UC_{ng,B} + P_{el,MCHP,out} \cdot UC_{el} + P_{el,grid,out} \cdot UC_{el} \quad (21)$$

where $UC_{ng,B}$ is the unit cost of natural gas supplied to the boiler of the reference system. The volumetric flow rate $\dot{V}_{ng,B}$ consumed by the boiler is calculated as follows:

$$\dot{V}_{ng,B} = P_{th,MCHP} / (\eta_B \cdot \rho_{ng} \cdot LHV_{ng}) \quad (22)$$

In Italy, the unit cost of natural gas for residential applications depends on the region of Italy where the natural gas is consumed, the level of cumulated natural gas consumption, and the application (cogenerative use or uses other than cogeneration) [43]. In addition it should be highlighted that several types of economic incentives are adopted by the Italian government for MCHP units to be financially feasible [44,45]:

- tax rebate on natural gas purchased;
- tradable white certificates, that involve achieving a mandatory energy-saving target against the “business-as-usual” scenario according to a specific table established by the Italian Regulatory Authority for Electricity and Gas [46];
- government capital grants on the purchase of the MCHP unit; according to the current legislation, it is equal to the 40% of the capital cost of the cogeneration device.

In this paper, a simplified analysis of the operating costs has been performed by assuming the following unit costs for natural gas and electricity according to the Italian scenario:

- an average unit cost of natural gas equal to 0.89 €/m³ [47] for applications other than cogeneration;
- an average unit cost of natural gas equal to 0.70 €/m³ [47] for cogenerative use;
- an average unit cost of electricity equal to 0.18 €/m³ [29].

Finally the comparison between the proposed and conventional systems is performed by assessing their environmental impact. With respect to this point, it should be highlighted that a comprehensive evaluation of micro-cogeneration technology should take into account also the impact due to the presence of plants spread over the territory that could increase the local pollution, in particular due to nitrogen oxides and carbon monoxide, and thus could worsen the local air quality [48]. A simplified approach is adopted in this paper by neglecting the local effects and performing the environmental comparison in terms of global carbon dioxide equivalent emissions:

$$\Delta CO_2 = (CO_2^{CS} - CO_2^{PS}) / CO_2^{CS} \quad (23)$$

The term CO_2^{PS} represents the carbon dioxide equivalent emissions associated to the proposed system, while the term CO_2^{CS} represents the carbon dioxide equivalent emissions associated to the system assumed as a reference. According to its definition, a positive value of ΔCO_2 indicates that the system based on micro-cogeneration technology allows to reduce the carbon dioxide emissions in comparison to the conventional system.

The assessment of the carbon dioxide equivalent emissions is carried out through an energy output-based emission factor approach [49]. According to this approach, the mass of a given pollutant x emitted while producing the energy output E can be worked out as:

$$m_x = \mu_x^E \cdot E \quad (24)$$

where μ_x^E is the energy output-based emission factor, that is, the specific emissions of x per unit of E . This factor depends upon several operating and structural variables, such as the specific equipment, partial load operation, age, state of maintenance, outdoor conditions, pollutant abatement systems, and so forth.

According to the values suggested in Refs. [50–52], in this study the CO_2 emission factor associated to the natural gas consumption is assumed equal to $200 \text{ gCO}_2/\text{kWh}_p$; this value represents the equivalent CO_2 emissions for a primary energy unit consumed by burning out natural gas.

The average CO_2 emissions for producing electricity depend on the technology mix adopted. Italy has a power generation system with no nuclear power and with about 20% hydro-electricity; the main part is thermal-based, with a mix of fuels from coal to oil. According to the values suggested in Refs. [50–52], in this work the CO_2 emission factor for electricity production is assumed equal to $525 \text{ gCO}_2/\text{kWh}_{el}$; this value represents the equivalent CO_2 emissions for an electric energy unit consumed with reference to the Italian scenario.

According to the above-mentioned approach, the values of CO_2^{PS} and CO_2^{CS} can be calculated by means of the following formulas:

$$\text{CO}_2^{\text{PS}} = P_{p,\text{MCHP}} \cdot \beta + P_{el,\text{grid,out}} \cdot \alpha \quad (25)$$

$$\text{CO}_2^{\text{CS}} = (P_{th,\text{MCHP}}/\eta_B) \cdot \beta + (P_{el,\text{MCHP,out}} + P_{el,\text{grid,out}}) \cdot \alpha \quad (26)$$

where α is the CO_2 emission factor for electricity production and β represents the CO_2 emission factor associated to the natural gas consumption.

6. Energy, economic and environmental analysis: methodology

Fig. 5a–c highlights the values of PES (Eq. (16)), ΔOC (Eq. (19)) and ΔCO_2 (Eq. (23)) calculated on the basis of the data measured during tests 1, 2, and 3, respectively. The values of PES, ΔOC and ΔCO_2 associated to the warm-up operation of the MCHP unit are not reported in these figures because they are extremely negative (up to around -250%) due to the fact that during that period the micro-cogeneration device consumes some fuel without producing electricity and/or heat. With respect to the remaining parts of the experiments, it can be noticed that:

- the values of PES range from -22% up to $+20\%$;
- the values of ΔOC range from 0% up to $+39\%$;
- the values of ΔCO_2 range from -20% up to $+23\%$;
- the largest values of PES, ΔOC and ΔCO_2 are achieved when both the cogeneration unit and electric chiller are activated;
- the lowest values of PES, ΔOC and ΔCO_2 are achieved during both cool-down and transient operation of the MCHP device.

The experimental results are analyzed in Fig. 5a–c based on the measured instantaneous power values. The same experiments are

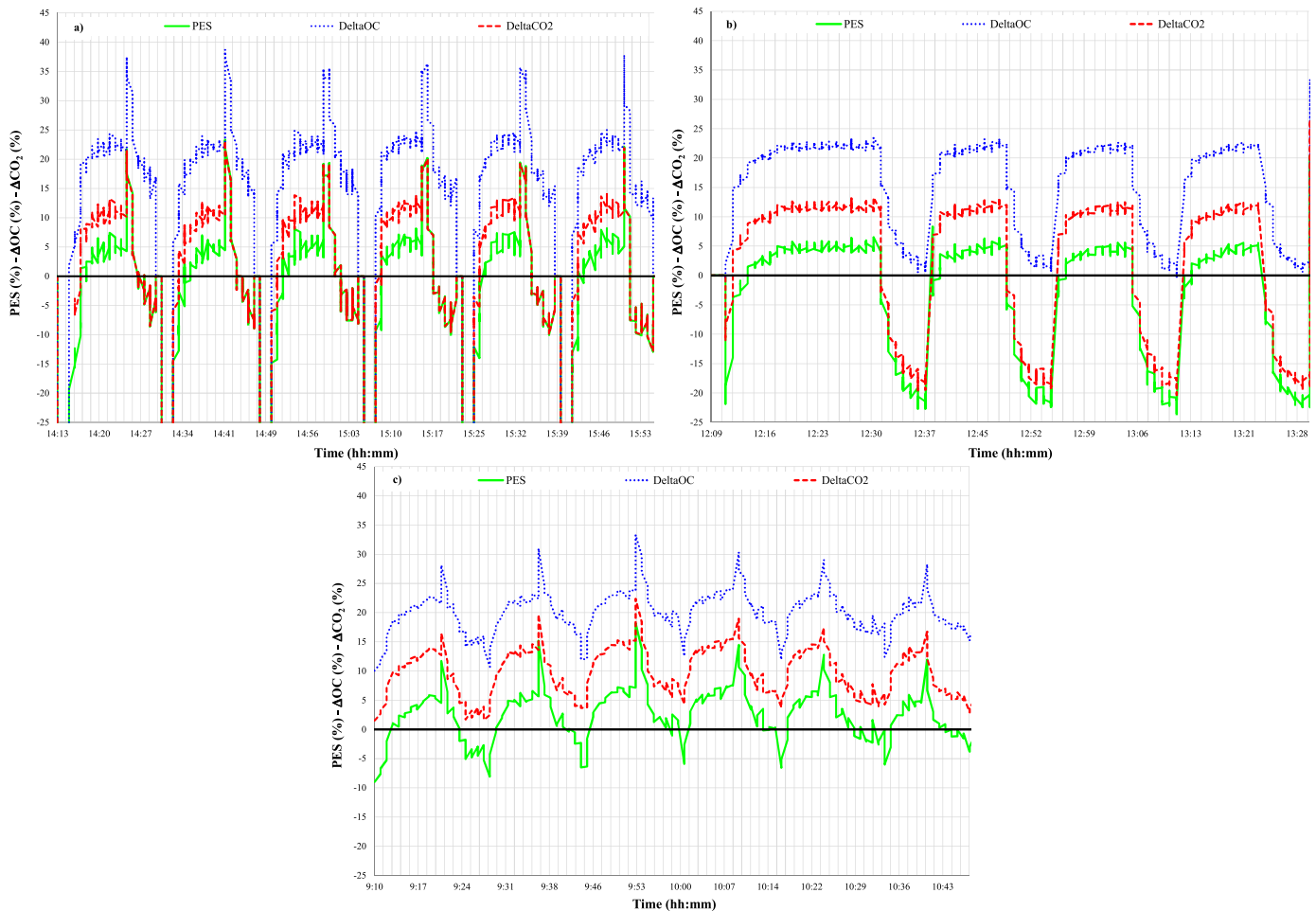


Fig. 5. Values of PES, ΔOC and ΔCO_2 calculated on the basis of the data measured during test 1 (a), test 2 (b) and test 3 (c).

described in terms of energy values in Table 5: this table reports the electric energy $E_{el,MCHP,out}$ supplied by the MCHP unit, the electric energy $E_{el,MCHP,in}$ consumed by the MCHP unit, the thermal energy $E_{th,MCHP}$ recovered from the MCHP device, the primary energy $E_{p,MCHP}$ consumed by the MCHP device, the electric energy $E_{el,grid,out}$ supplied by the grid, the electric energy $E_{el,ACWC}$ consumed by the ACWC, the cooling energy $E_{cool,ACWC}$ supplied by the ACWC and the electric energy $E_{el,appliances}$ consumed by the electric appliances (pumps, lighting, PCs, printers, plotters, auxiliaries) measured during the whole duration of tests 1, 2 and 3; the values of PES (Eq. (16)), ΔOC (Eq. (19)) and ΔCO_2 (Eq. (23)) calculated on the basis of the experimental energy flows are also calculated in the same table.

The data reported in Table 5 highlight that, if compared to the conventional system, the proposed system:

- allows to reduce the operating costs ($\Delta OC > 0$), whatever the experimental test under consideration is;
- is able to decrease the equivalent carbon dioxide emissions ($\Delta CO_2 > 0$) under the operating conditions associated to both tests 2 and 3;
- produces a reduction of primary energy consumption ($PES > 0$) only in the case of test 3.

Tests 2 and 3 show better results than those associated to test 1 mainly thanks to the facts that both the average electric load is larger (allowing a greater electric efficiency of the MCHP unit) and the time spent under warm-up/cool-down operation is almost negligible.

Table 6 shows the duration of the period with simultaneous operation of both MCHP unit and ACWC for each experiment (the cogeneration unit and/or the chiller are switched off during the remaining part of the tests); the fraction of time characterized by simultaneous operation of both the MCHP device and ACWC with positive values of PES, ΔOC and ΔCO_2 is also calculated in Table 6. This table shows that, in comparison to the reference system, simultaneous operation of both MCHP unit and ACWC always reduces the operating costs; the possibility to decrease both carbon dioxide equivalent emissions and primary energy consumption depends on the operating conditions: the better results are achieved when the average electric load sensed by the MCHP unit is large and/or transient operation associated to warm-up/cool-down of the micro-cogeneration device is minimized.

As above highlighted, thermal energy recovered from micro-cogeneration devices can be profitably exploited in residential applications during summer for domestic hot water production and/or activating absorption/adsorption systems for space cooling purposes. With respect to this point, it should be noticed that the values of PES (Eq. (16)), ΔOC (Eq. (19)) and ΔCO_2 (Eq. (23)) reported in Fig. 5 and Tables 5 and 6 are calculated by assuming that 100% of

Table 6

Energy, economic and environmental analyses of simultaneous operation of both MCHP unit and ACWC.

	Test 1	Test 2	Test 3
Time of simultaneous operation of both MCHP unit and ACWC (min)	53	53	19
Time percentage of simultaneous operation of both MCHP unit and ACWC with $PES > 0$ (%)	78	88	82
Time percentage of simultaneous operation of both MCHP unit and ACWC with $\Delta OC > 0$ (%)	100	99	100
Time percentage of simultaneous operation of both MCHP unit and ACWC with $\Delta CO_2 > 0$ (%)	86	98	100

the thermal output of the micro-cogeneration device is effectively used: in the case of the heat recovered from the MCHP unit is not fully exploited, the results of the energy, economic and environmental analyses could drastically change. In order to better highlight that micro-trigeneration deployment is economically, energetically and environmentally affected by the opportunity to recover the cogenerated heat during summer, Fig. 6a–c highlights the variation of PES (Eq. (16)), ΔCO (Eq. (19)) and ΔCO_2 (Eq. (23)) with the fraction of $E_{th,MCHP}$ effectively used; the values reported in these figures are calculated on the basis of the data measured during tests 1, 2 and 3.

Fig. 6a–c shows that the potential savings associated to the utilization of the system under investigation significantly reduce at increasing the portion of thermal energy recovered from the MCHP unit that is not effectively used. In particular, these figures highlight that this reduction is maximum for the values of PES and minimum for the values of ΔCO_2 ; as example, it can be noticed that in the case of the percentage of $E_{th,MCHP}$ effectively exploited reduces from 100% to 50% under the operating conditions of test 3:

- ✓ the value of PES reduces from +2.5% to –34.3%;
- ✓ the value of ΔCO reduces from +20.0% to –11.7%;
- ✓ the value of ΔCO_2 reduces from 10.1% to –19.9%.

Finally it should be highlighted that the temperature of water within the tank connected to the MCHP system increases in the case of the cogenerated heat is only partially exploited and this could activate the internal radiator of the cogeneration unit for thermal energy dissipation: such condition should be strictly avoided as it represents a condition in which recovered heat is very low and the performance of the micro-trigeneration system are quite poor.

7. Conclusions

In the present paper the performance of a micro-trigeneration system were experimentally evaluated during three typical days of summer operation and compared with those of a conventional system based on separate energy production from energy, economic and environmental point of views.

The comparison highlighted that the micro-trigeneration system is potentially able to save primary energy up to around 20%, reduce the operating costs up to around 39%, and decrease the carbon dioxide equivalent emissions up to around 23%. The largest savings can be achieved in the case of the average electric load is high and/or the transient operation of the cogeneration device is limited; the proposed system showed the worst performance mainly during both warm-up and cool-down of the MCHP unit. The potential savings are strongly related to the fraction of cogenerated

Table 5

Comparison between proposed and reference systems based on measured energy flows.

	Test 1	Test 2	Test 3
$E_{el,MCHP,out}$ (kJ)	7448.4	10304.7	19967.1
$E_{el,MCHP,in}$ (kJ)	548.7	19.5	0
$E_{th,MCHP}$ (kJ)	39962.6	37439.5	52292.9
$E_{p,MCHP}$ (kJ)	68577.6	64736.6	98774.4
$E_{el,grid,out}$ (kJ)	3753.5	1850.1	2088.3
$E_{el,ACWC}$ (kJ)	10646.5	8602.2	8445.6
$E_{cool,ACWC}$ (kJ)	23070.8	18514.1	18242.3
$E_{el,appliances}$ (kJ)	164.2	3533.1	13609.8
PES (%)	–11.7	–1.2	2.5
ΔOC (%)	8.1	17.2	20.0
ΔCO_2 (%)	–6.3	5.3	10.1

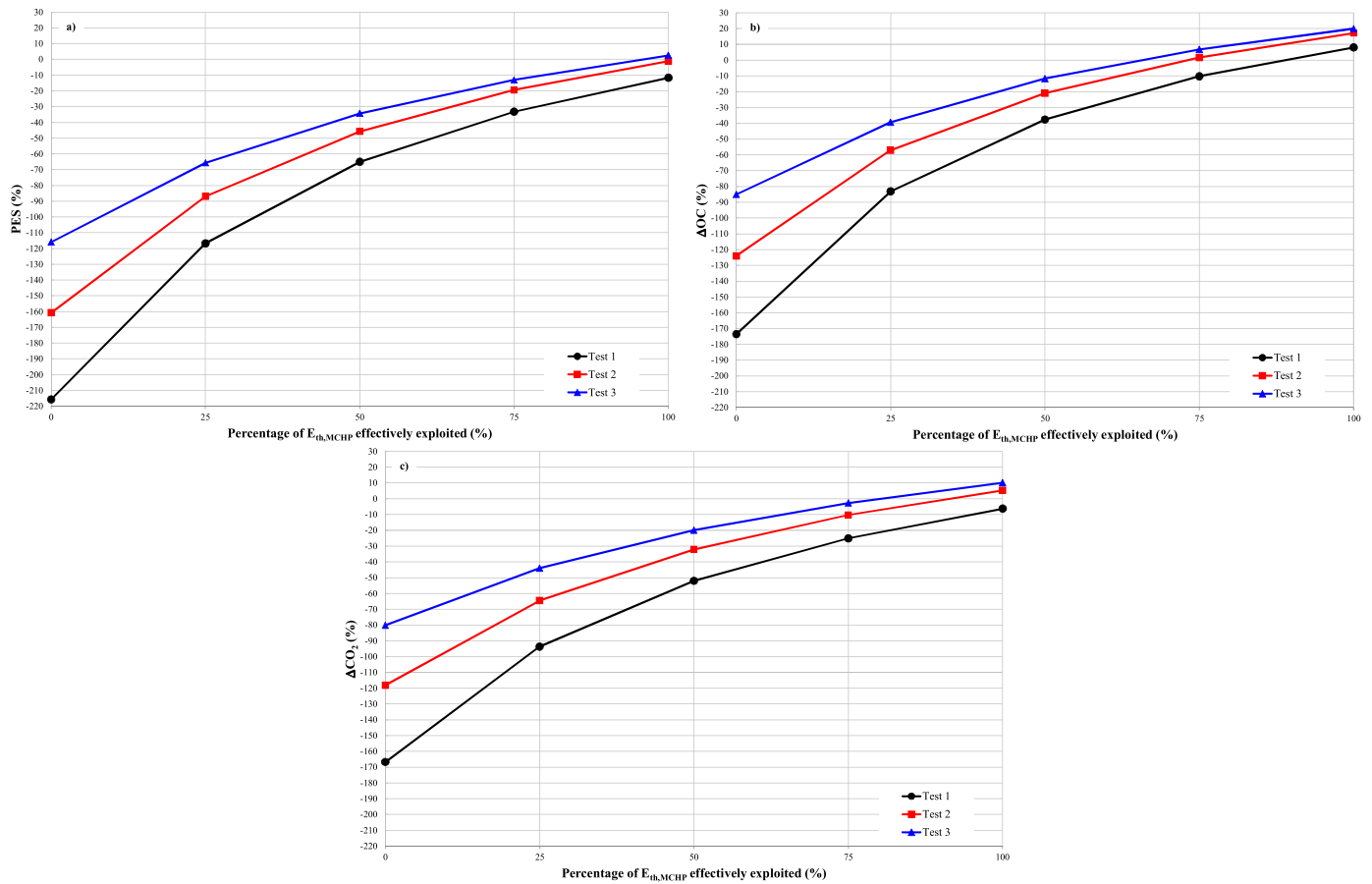


Fig. 6. Values of PES (a), ΔOC (b) and ΔCO_2 (c) as a function of the fraction of $E_{th,MCHP}$ effectively used during tests 1, 2 and 3.

heat that is effectively exploited for domestic hot water production and/or activating absorption/adsorption systems: the energy, environmental and economic advantages significantly reduce upon decreasing this fraction.

Nomenclature

Latin letters

ACWC	air-cooled water chiller
c	specific heat (kJ/kg K)
CO_2	carbon dioxide equivalent emission (kg CO_2)
COP	coefficient of performance
E	energy (kJ)
FC	fan-coil
IHE	internal heat exchanger
HWS	hot water storage
LHV	lower heating value (kJ/kg)
m	mass of a given pollutant (kg)
M	water volumetric flow meter
MCHP	micro combined heat and power
MCCHP	micro combined cooling heat and power
MG	natural gas volumetric flow meter
OC	operating cost (€)
P	power (kW)/pump
PES	primary energy saving (%)
R	electric resistance
T	temperature/resistance thermometer

u	combined standard uncertainty (%)
UC_{ng}	unit cost of natural gas (€/Nm ³)
UC_{el}	unit cost of electric energy (€/kWh _{el})
$\dot{V}$	volumetric flow rate (m ³ /s)
W	wattmeter

Greeks

α	CO_2 emission factor for electricity production (g CO_2 /kWh _{el})
β	CO_2 emission factor associated to the natural gas consumption (g CO_2 /kWh _p)
Δ	difference
η	efficiency (%)
μ	energy output-based emission factor (g CO_2 /kWh)
ρ	density (kg/m ³)

Superscripts

CS	conventional system
E	energy
PS	proposed system

Subscripts

ACWC	air-cooled water chiller
appliances	pumps, lighting, PCs, printers, plotters, auxiliaries
air	air
B	boiler
cool	cooling
cw	cooling water

el	electric
grid	Italian electric grid
HWS	hot water storage
in	inlet
MCHP	micro combined heat and power
ng	natural gas
out	outlet
p	primary
PP	power plant
th	thermal
w	water
x	generic pollutant

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